SPECIAL REPORT

A THEORETICAL TECHNIQUE FOR PREDICTING THE RELIABILITY OF SOLENOID VALVES

January 1962

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under Contract SD-77

ARINC RESEARCH CORPORATION
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Special Report

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FOREWORD

This is a special technical report prepared by ARINC Research Corporation for the Advanced Research Projects Agency under Contract SD-77.

This report describes a technique for determining the reliability of the solenoid valves used in satellite attitude control systems. Although ARINC Research considers this preliminary effort as a step in the right direction for predicting the reliability of mechanical and electromechanical devices, it suggests that a detailed empirical analyses be undertaken in the near future to more definitely confirm both the theoretical modes of failure analysis and the theoretical prediction technique presented herein.

TABLE OF CONTENTS

		Page
1.	INTRODUCTION	1
	1.1 Background	1
	1.2 Purpose and Scope of Report	1
2.	FACTORS AFFECTING SOLENOID VALVE RELIABILITY	5
	2.1 Direct Lift vs. Pilot-Operated Valves	5
	2.2 Factors Affecting Poppet and Seat Reliability	9
	2.2.1 Response Time	9
	2.2.2 Impact Stress	12
	2.2.3 Repeated Stress	18
3.	RELIABILITY ESTIMATE FROM IMPACT STRESS AND THE EFFECTS OF REPEATED STRESS	21
4.	THE VARIABLES AFFECTING THE RELIABILITY OF THE SEAT AND POPPET	23
	4.1 Poppet Velocity	23
	4.2 Poppet Travel	24
	4.3 Temperature vs. Reliability	25
	4.4 Static Load	25
	4.5 Space Application	27
	4.5.1 Precycling	27
	4.5.2 Materials	29
5.	COMBINED RELIABILITY	31
	5.1 Statistical Aspect	31
	5.2 Part Reliabilities	31
	5.2.1 Solenoid and Spring	31
	5.2.2 Poppet and Seat	32
6.	CONCLUSIONS AND RECOMMENDATIONS	35
	6.1 Conclusions	35
	6.2 Recommendations	35

LIST OF FIGURES AND TABLES

Figure		Page
1	Typical Configuration of a Cold Gas Attitude Control System	2
2	Typical Attitude Control Axes for Control Jets	5
3	Typical Direct Lift Solenoid Valve	6
4	Poppet and Seat Combinations Generally Used in Solenoid Valves	8
5	Valve Poppet and Seat Nomenclature for Response Time and Stress Calculations	13
6	Work, Force and Deflection Relationship Between the Poppet and Seat of a Solenoid Valve	15
7	Conventional S-N Diagram for the Relationship Between Unit Stress and Number of Stress Cycles	19
8	S-N Curve Obtained from Limited Tests of a Typical Group of 2024-T4 Specimers Axially Loaded, Stressed in Compression	22
9	Response Time vs. Cycles to Failure of a Typical Solenoid Valve Design with all Other Variables Constant	24
10	Poppet Travel vs. Cycles to Failure of a Typical Solenoid Valve Design with all Other Variables Constant	25
11	The Effects of Temperature Change on a Material in Repeated Stress	26
12	Trade-Offs Affecting Reliability, Leak Rate, and Load	26
13	Average Solenoid Valve Leakage Rate vs. Number of Cycles with Metal to Metal Sealing, Average of Various Inlet Pressures, and Approximate 0 PSIA Outlet Pressure	28
14	Average Solenoid Valve Leakage Rate vs. Number of Cycles, with Nonmetal Sealing, Average of Various Inlet Pressure, and Approximate 0 PSIA Outlet Pressure	28

LIST OF FIGURES AND TABLES (continued)

Figures		Page
15	Theoretical Evaporation of Some Metals in a Vacuum at Approximately 10^{-9} to 10^{-10} Millimeters of Mercury	29
16	Weight Losses for Some Nonmetals in a Vacuum at Approximately 10^{-9} to 10^{-10} Millimeters of Mercury	30
17	Reliability Considering Only Part Failures by an Exponential Distribution (Springs, Solenoids)	33
18	Reliability Considering Only Part Failures Described by a Normal Distribution (Poppet and Seat)	34
19	Combining of Part Reliabilities to Give Solenoid Valve Reliability	34

Table		
1	Estimated Failure Rates and Mean Lives of Solenoid Valve Parts	32

1. INTRODUCTION

1.1 Background

With the advent of larger and more sophisticated satellites, the control of their attitudes becomes an increasingly important area requiring application of design techniques that will assure the highest probability of a successful mission. For instance, the success of a photographic surveillance satellite depends to a large extent upon the proper functioning of the attitude control system because "out-of-tolerance" perturbations result in poor quality photographs. The accuracy achieved is both a function of the response time and the ability of the control system to operate intermittently over an extended period of time.

A study of the past history of attitude control systems reveals that the demands imposed on attitude control are becoming progressively more stringent. The design of attitude control systems has progressed in complexity to a point where two complete systems are now required for proper control. One system, referred to as "Vernier" control change, controls small degrees of attitude position; another which controls large degrees of attitude position change, is referred to as "coarse" control.

The "Vernier" control system is usually comprised of a system of reaction wheels employing the reaction effect of an inertia wheel. The "coarse" control system generally consists of a cold gas reaction system employing the reactive forces of gas, expanded through De Laval-type nozzles.

1.2 Purpose and Scope of Report

Since the success of present day satellite missions depends heavily upon the reliability of the attitude control system, ARINC Research Corporation initiated a study for the purpose of establishing a theoretical prediction technique that is applicable to solenoid valves.

The theoretical approach was chosen because of the dearth of available reliability information. Some of the scant reliability data were used by ARINC Research from time to time for comparison checks between the results obtained theoretically and the empirical data. However, neither the data nor the comparison results warranted much confidence.

It is realized that the pressure regulator plays an important role in a control system. However, a superficial study of attitude control system-failure modes indicated that the gas solenoid valve would more than likely present the most problems, because its reliability is assumed to depend almost entirely upon the ability of the solenoid valves to contain the reaction gas supply from the outside environment. Failure of any one of the six solenoid valves to close would constitute a system failure since the stored gas would be depleted before the planned mission is accomplished. Figure 1 is a typical configuration of a cold gas attitude control system.

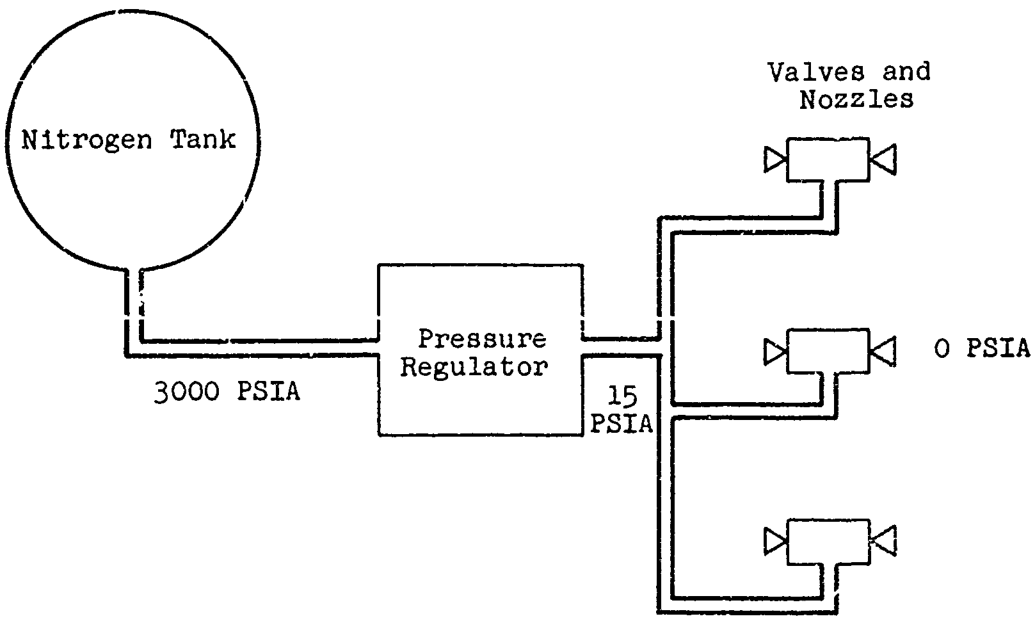


FIGURE 1

TYPICAL CONFIGURATION OF A COLD GAS ATTITUDE CONTROL SYSTEM

Therefore, the primary oojective of this study was to perform a theoretical analysis of solenoid valve reliability and to develop a theoretical prediction method by which solenoid valve reliability could be determined. A secondary objective concerned suggestions on the proper reliability application of solenoid valves for the attitude control of satellites.

2. FACTORS AFFECTING SOLENOID VALVE RELIABILITY

2.1 Direct Lift vs. Pilot-Operated Valves

For long orbit life requirements, the attitude control solenoid-valves will operate a large number of times while performing the stabilizing function for the satellite, which tends to oscillate continuously about its control axes. Figure 2 is a schematic which shows the control axes of a satellite.

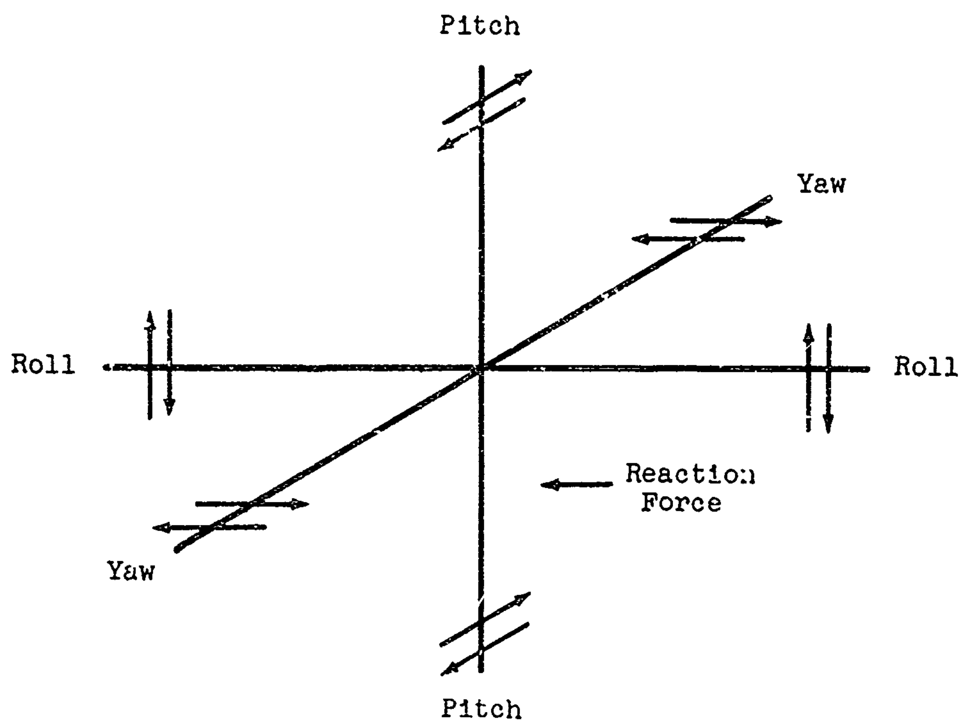


FIGURE 2

TYPICAL ATTITUDE CONTROL AXES FOR CONTROL JETS

The gas-control valves used for satellite attitude-control are generally direct-lift valves using the electromechanical force of a solenoid to actuate the valve. Occasionally the pilot-operated valve is employed, although it is designed primarily for high-flow rate and high-pressure differential applications. Therefore, because most cold gas reaction attitude-control systems employ low-flow rate and low-pressure differential valves, this study concerns only the direct-lift valve which is designed primarily for such applications. The results obtained from a study of direct-lift valves can also be applied to the pilot-operated valves since, in each type, the modes of failure are identical.

A direct-lift solenoid valve, as shown in Figure 3, is composed of the following components: body; valve seat; poppet with integral armature; solenoid for the activating force; return spring.

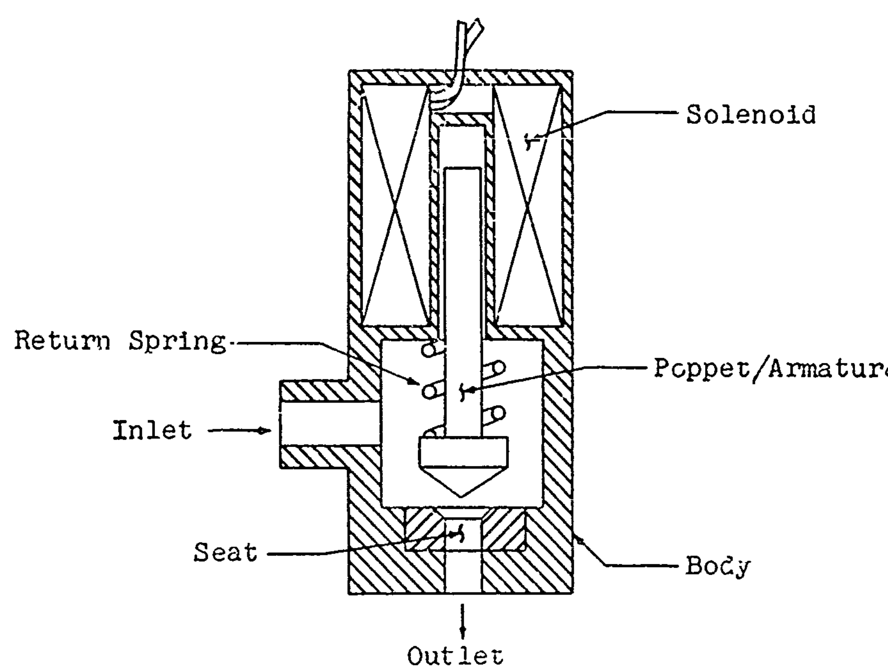


FIGURE 3

TYPICAL DIRECT LIFT SOLENOID VALVE

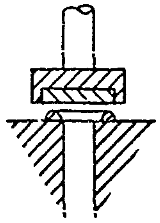
Although there are many different types of direct lift valves, they were considered to be identical for the purposes of this reliability study since each contains a poppet and seat, which components are the main reliability consideration at this time. Figure 4 shows a few of the predominate poppet and seat combinations used in valves, and various combinations of poppet and seat arrangements used for effective sealing. Figure 4(a) illustrates a metal-to-nonmetal sealing surface; figures 4(b) through 4(f) illustrate metal-to-metal sealing surfaces.

It is difficult, if not impossible, to analyze the potential reliability level of each poppet and seat combination shown. However, there are combinations which will exhibit a higher reliability for a specified application and which should be carefully considered. For example: the "insert seat," if properly installed, will allow the body of the valve to be constructed of a light-weight metal, such as aluminum, which has low-strength properties, and the seat to be made of some tough, durable, high-strength alloy steel. Another example would be use of the "conical poppet" for a high-shock environment as it affords better alignment between the poppet and seat.

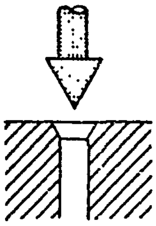
When a gas reaction-control system is applied to the attitude control of a satellite, the design of the system must be such that the reaction gas-supply can meet the mission requirements within close limits. An oversupply of gas would impose problems of both weight and size upon the satellite. Thus, an early depletion of the reaction gas would result in mission failure for a satellite that depends on attitude control through cold-gas reaction for mission accomplishment.

When not in use, the solenoid valve is required to seal the gas supply in the attitude control system against the vacuum conditions of outer space, as shown in Figure 1. It is also required to operate a countless number of cycles while stabilizing the satellite during long mission times. It, therefore, is a major reliability item in the attitude control system.

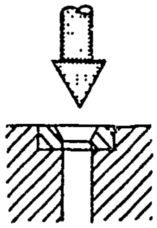
The theoretical application of the solenoid valve for attitude control of a satellite may appear to be quite simple at first glance. However, because it imposes many reliability problems which are difficult to analyze, the converse is true. The major difficulty stems from the fact that most solenoid valve manufacturers avoid the challenge



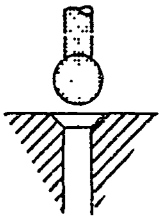
(a) Raised Seat, Insert Poppet



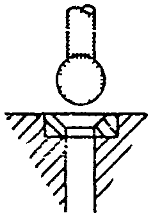
(b) Conical Poppet, Chamfered Seat



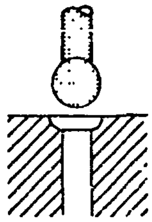
(c) Conical Poppet, Insert Seat



(d) Spherical Poppet, Chamfered Seat



(e) Spherical Poppet, Insert Seat



(f) Spherical Poppet, Spherical Seat

FIGURE 4

POPPET AND SEAT COMBINATIONS GENERALLY USED IN SOLENOID VALVES

of manufacturing a valve which will meet a longevity specification, thus limiting the compilation of data on solenoid valve failure. The situation is further complicated by the unforeseen problems of outer space environment.

The majority of the manufacturers contacted by ARINC Research for information pertaining to this study could only guess the expected life of solenoid valves. The reason for this is their reluctance to conduct tests of sufficient duration to establish the level of reliability attained after extended cycles of operation.

At the beginning of this study, all solenoid valves were expected to exhibit a typical, or average, expected life. This hypothesis was negated by the data which were compiled and evaluated. They indicated that the life of a solenoid valve depends to a large extent on its particular application.

As a result of the preliminary study, the derivation of a set of theoretical equations was deemed necessary, based on general operating conditions that would apply to any solenoid valve application for calculating the expected life of the valve.

Limited data obtained from various manufacturers have tentatively supported the assumption that the life of a solenoid valve primarily depends upon the poppet and seat. As a given probability exists that the solenoid and the return spring will fail during the life of the valve, it is necessary to theoretically analyze the reliability aspects of the poppet and seat, using empirical failure-data for the solenoid and spring.

2.2 Factors Affecting Poppet and Seat Reliability

Preliminary investigations indicated that the life of a poppet and seat are affected by three fundamental variables, namely: response time; impact stress; and repeated stress.

2.2.1 Response Time

The response time is the first major aspect affecting the reliability of a solenoid valve. It can be defined as: the time required to open, upon activation of the solenoid, and the time required to close, upon deactivation of the solenoid. The response time of a solenoid valve is a function of the following variables:

(1) mass of moving parts,
(2) pressure differential,
(3) armature travel,
(4) coil winding,
(5) applied current,
(6) friction of moving parts.

The response time can be described by the following elementary dynamic equations.

$$F = \frac{W}{g} a \qquad \text{Equation (1)}$$

where

$$a = \frac{2d}{t^2}$$

d = distance traveled (feet)

t = time (seconds)

W = weight (pounds)

g = acceleration due to gravity (feet per second, squared)

F = resultant force (pounds)

Then, by substitution and transposition, the response time is

$$t = 1.4 \left(\frac{Wd}{Fg} \right)^{1/2} \qquad \text{Equation (2)}$$

Applying equation (2) to the general case of direct-lift solenoid valves, as shown in Figure 3, an approximation of the response time -- open and close -- can be made if friction, buildup and decay time of magnetic flux in the solenoid, and flow turbulence are not considered.

Response time to open is given by the equation:

$$t_o = 1.4 \left(\frac{W_o\ d}{F_o\ g} \right)^{1/2} \qquad \text{Equation (3)}$$

where

W_o = (weight of the armature + spring preload + force of pressure differential on the orifice area)

d = distance traveled by armature

F_o = magnetic force of pull-in of the solenoid.

Response time to close is given by the equation:

$$t_c = 1.4 \left(\frac{W_c\ d}{F_c\ g} \right)^{1/2} \qquad \text{Equation (4)}$$

where

W_c = weight of the armature

d = distance traveled by armature

F_c = spring force + dynamic force of gas on piston.

Equations (3) and (4) were employed to calculate the response time of a number of solenoid valves of different types, and the calculations were compared with corresponding test results. The results of the equations proved to be accurate within $\pm$ 3 milliseconds. When using those equations, the dynamic reaction of the flow of gas through the valve was assumed to equal zero. Therefore, were the dynamic reactions considered, it is hypothesized that the results obtained would have been in closer agreement with the actual test data.

Since the instantaneous velocity $(v)^o$ is equal to the acceleration (a) multiplied by the time (t), the above discussion emphasizes the fact that response time is a critical function in the life of solenoid valve. The effects of response time on reliability are discussed in more detail in Section 4.

2.2.2 Impact Stress

It is common knowledge that loads applied suddenly and with a finite velocity, produce much greater stresses than if applied gradually. Therefore, it is theorized, the impact force between the seat and poppet produces the major stress in a solenoid valve.

A mathematical analysis of the impact stress between the seat and poppet of a solenoid valve follows.

Dimensional Terms

W_p = weight of the poppet, lb.

A_o = orifice area* in.^{2}

A = seating surface normal to the impinging forces*, in.^{2}

F_s = spring preload at the closed position*, lb.

F_p = pressure differential* $(P_u - P_o)$ times orifice area* (A_o), lb.

g = acceleration due to gravity, ft/sec^2

Δ = deflection of the materials, ft.

v = velocity of the poppet and instant of contact which is equal to the distance* (d) traveled by the poppet from the open to the closed position multiplied by 2 and divided by the response time (t) to close $(d = 1/2\ vt)$, ft/sec.

E = modulus of elasticity, lb/in^2

ϵ = strain, in/in

L = theoretical depth of the poppet and seat affected by the impacting force, ft.

S = impact stress, lb/in^2

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* See Figure 5

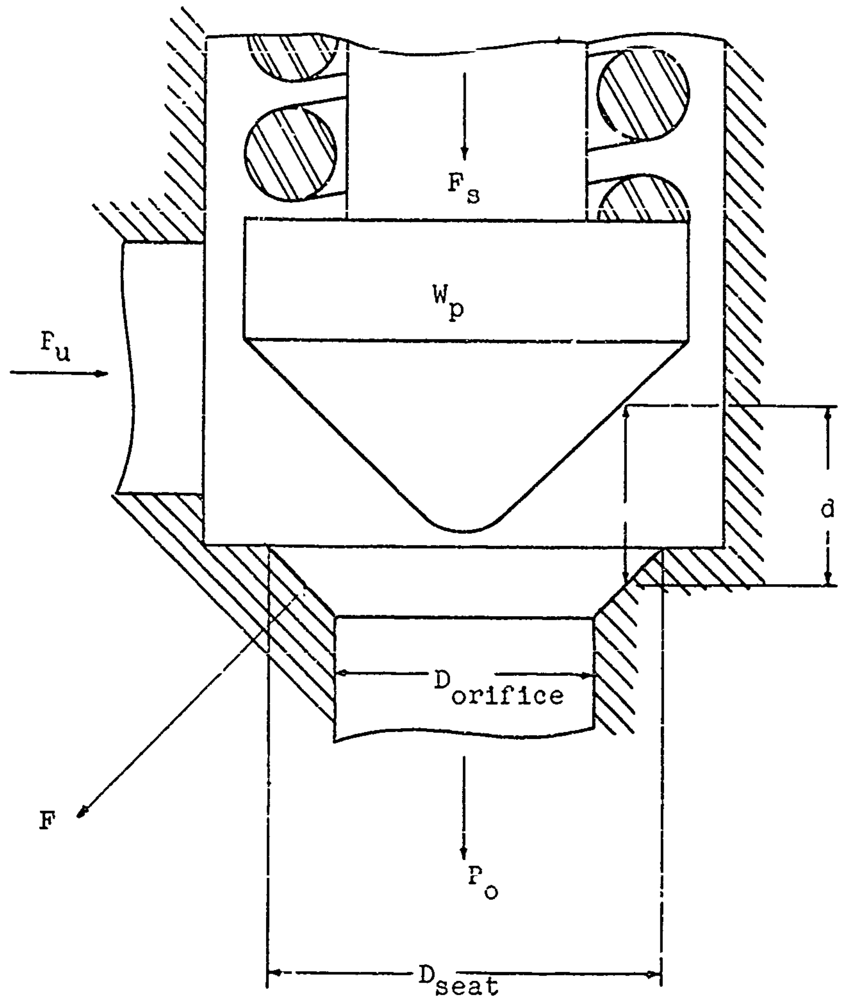

FIGURE 5

VALVE POPPET AND SEAT NOMENCLATURE
FOR RESPONSE TIME AND STRESS CALCULATIONS

A material, when resisting an impact load, is assumed to act in the same way as when resisting a gradually applied (static) load; in other words, stress is proportional to strain until the proportional limit is reached. Therefore, the work (U) absorbed by a material may be expressed as a force (F) times the total deformation Δ_T (see Figure 6a, illustrating that work equals the area OAB up to the proportional limit).

Excluding the resultant work performed by the spring during the expansion through a deflection (Δ_T), and assuming a linear poppet-velocity closing (v), the equation for the resultant work (U) done by the weight of the poppet (W_p), the spring preload (F_s), and the pressure force (F_p), through a deflection (Δ_T) can be written as follows:

$$U = \frac{W_p v^2}{2g} + \frac{F_s \Delta_T}{2} + \frac{F_p \Delta_T}{2}$$

When equilibrium is established, the poppet and seat will absorb the resultant work (U) which is equal to one-half the product of the resisting force (F) and the deflection (Δ_T). Therefore

$$\frac{W_p v^2}{2g} + \frac{F_s \Delta_T}{2} + \frac{F_p \Delta_T}{2} = \frac{F\Delta_T}{2}$$

or

$$\frac{W_p v^2}{g} + \Delta_T(F_s + F_p) = F\Delta_T \qquad \text{Equation (5)}$$

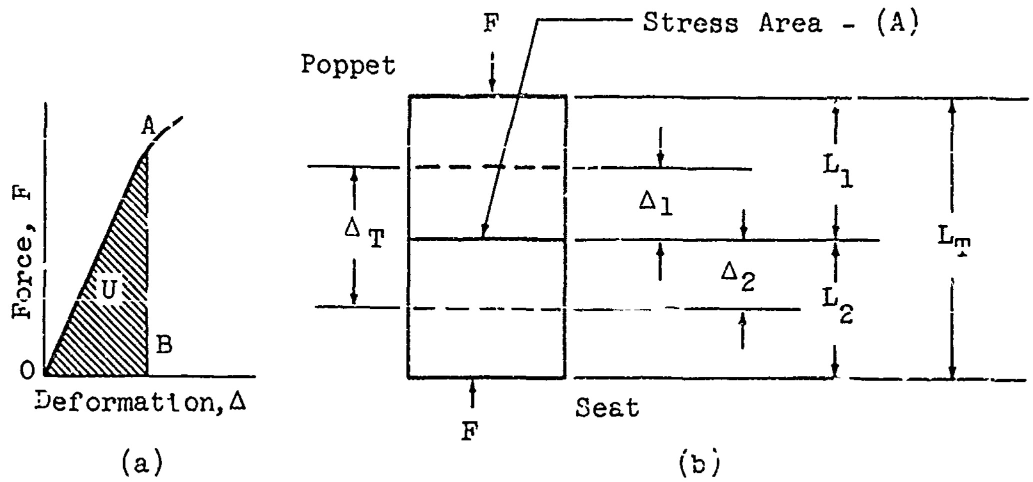

FIGURE 6

WORK, FORCE, AND DEFLECTION RELATIONSHIP BETWEEN THE POPPET AND SEAT OF A SOLENOID VALVE

The stress areas are assumed to be equal and, from the general relationships, stress (S) equals the force (F) divided by the area (A), then

$$S\Delta_T = \frac{F\Delta_T}{A}$$

or

$$F\Delta_T = \Delta_T SA$$

Then equation (5) becomes

$$\frac{W_p v^2}{g} + \Delta_T(F_s + F_p) = \Delta_T SA$$

or

$$\frac{W_p v^2}{g\Delta_T A} + \frac{F_s + F_p}{A} = S \qquad \text{Equation (6)}$$

The deflection shown in Figure 6(b), (Δ_T) is equal to the deflection in the seat (Δ_2) plus the deflection in the poppet (Δ_1); therefore

$$\Delta_T = \Delta_1 + \Delta_2 \qquad \text{Equation (7)}$$

Assuming the stress in the poppet and seat to be equal, $S_1 = S_2 = S$; then from the relationship $\Delta_T = SL/E$, equation (7) becomes

$$\Delta_T = \frac{S(L_1E_2 + L_2E_1)}{E_1E_2} \qquad \text{Equation (8)}$$

substituting equation (8) in equation (6), then

$$\frac{W_p v^2 E_1 E_2}{gAS(L_1E_2 + L_2E_1)} + \frac{F_s + F_p}{A} = S$$

or

$$S^2 = \frac{W_p v^2 E_1 E_2}{gA(L_1E_2+L_2E_1)} + \frac{S(F_s+F_p)}{A} \qquad \text{Equation (9)}$$

Assuming the following:

$$\frac{F_s+F_p}{A} = C_1 \qquad \text{and} \qquad \frac{W_p v^2 E_1 E_2}{gA(L_1E_2+L_2E_1)} = C_2$$

Substituting C_1 and C_2 and equating to zero, equation (9) becomes

$$S^2 - C_1S - C_2 = 0 \qquad \text{Equation (10)}$$

From the quadratic formula, the impact stress (S) in equation (10) is

$$S = \frac{1}{2}\left[C_1 \pm (C_1^2 + 4C_2)^{1/2}\right] \qquad \text{Equation (11)}$$

The maximum stress is to be found and, upon substituting the equivalents for C_1 and C_2 in equation (11), the impact stress (S) is equal to

$$S = \frac{1}{2}\left\{\frac{F_s+F_p}{A} + \left[\left(\frac{F_s+F_p}{A}\right)^2 + \frac{4W_p v^2 E_1 E_2}{gA(L_1E_2+L_2E_1)}\right]^{1/2}\right\} \qquad \text{Equation (12)}$$

If the materials used in the poppet and seat are identical, then $E_1 = E_2$, and equation (12) becomes

$$S = \frac{1}{2}\left\{\frac{F_s+F_p}{A} + \left[\left(\frac{F_s+F_p}{A}\right)^2 + \frac{4W_p v^2 E}{gA(L_1+L_2)}\right]^{1/2}\right\}$$

Since equation (12) has the lengths L_1 and L_2 for the depth of the poppet and seat effected by the impact force, the degree of confidence placed in equation (12) depends largely on the accuracy with which the lengths L_1 and L_2 were determined. As accurate approximations of L_1 and L_2 require extensive stress analyses, the following assumptions were made to determine L_1 and L_2 in order to stay within the boundaries of this study:

1. The following relationships, employing basic stress analysis equations in association with Figure 6(b), have been established.

   $$L_T = L_1 + L_2$$

   $$\Delta_T = \Delta_1 + \Delta_2$$

   $$S_1 = S_2 = S$$

   $$\epsilon_T = \epsilon_1 + \epsilon_2$$

2. Using equation (6) and substituting the lowest exhibited yield-stress of the materials involved as the impact stress (S), an approximate deflection (Δ_T) can be calculated.

3. Substituting the calculated (Δ_T), from Item 2, and the equivalents given in Item 1, in the general relationships $L_T = \Delta_T/\epsilon_T$ then,

   $$L_T = L_1 + L_2 = \frac{\Delta_T}{\epsilon_1 + \epsilon_2}$$

4. For most cases L_1 can be assumed to equal L_2, then

   $$L_1 = L_2 = \frac{\Delta_T}{2(\epsilon_1 + \epsilon_2)}$$

Since the stress area is assumed to be the seating surface between the poppet and seat, it is important to point out that F_s, F_p, and W_p must be calculated for the surface normal to the applied force.

2.2.3 Repeated Stress

Since the solenoid valve in an attitude control system for a satellite will require many cycles of operation, the effect of a repeated load on a material must be defined.

A repeated load is a force that is applied many times to a member causing continually varying stress in the material, usually through some definite range. If a stress is developed in a member and then released, the member is said to have been subjected to a cycle of stress.

Stress analysis literature often refers to the fatigue strength of a material as its ability to resist repeated stress. Early in the study of strength of materials (especially metals), analysis of the empirical data revealed that members usually failed (fractured) under repeated loads of much smaller magnitude than static loads that caused failure. The mode of failure of a member caused by repeated loads can be classified as a gradual or progressive fracture. The fracture appears to start at a point where the stress is highest -- the center of a concentrated or highly localized stress. When the load is repeated enough times, a small crack appears and gradually spreads until the member ruptures without measurable yielding as a whole.

The tendency of materials to fracture under repeated stress conditions increases with the maximum stress applied and with the range of stress. Stressing a material from zero to maximum stress is more destructive than stressing, say, from one-half maximum stress to maximum stress. For any given stress thus produced above a certain critical value, a material will fracture after a sufficient number of cycles. As the applied stress is decreased, the number of cycles required to produce failure increases; and, when the stress is equal to or less than the critical value mentioned, the material appears to be able to withstand an indefinitely large number of cycles. This critical stress is the endurance limit.

The behavior of a material under repeated stress is well represented by the so-called S-N diagram, which is a graph plotted with the unit stress(s), as ordinates, and the number of cycles (n) producing fracture, as abscissas. Such a graph, as shown in Figure 7, starts at the stress corresponding to the static ultimate strength and slopes downward until it becomes horizontal at the endurance limit.

The transition may be abrupt, in which case the endurance limit is sharply defined; or it may be gradual, in which case the endurance limit may be vaguely defined or even nonexistent. The number of stress cycles necessary to establish the endurance limit varies according to the material.

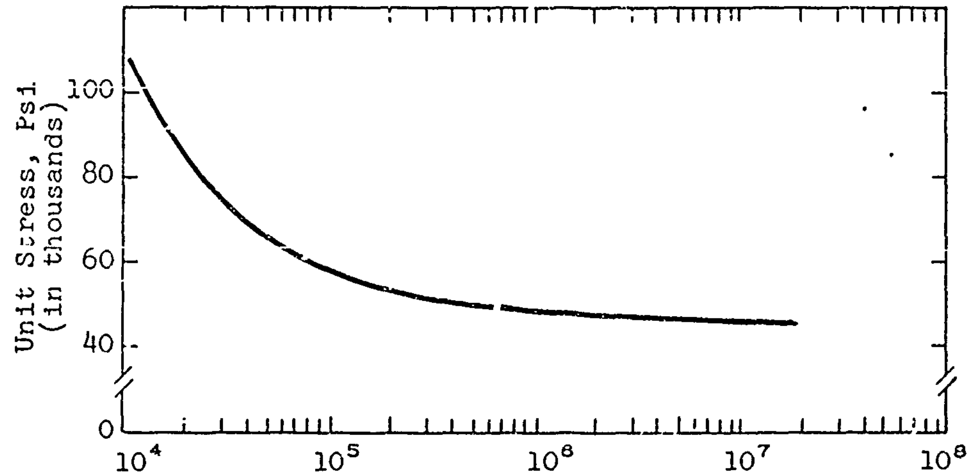

FIGURE 7

CONVENTIONAL S-N DIAGRAM FOR THE RELATIONSHIP BETWEEN UNIT STRESS AND NUMBER OF STRESS CYCLES

Some metals, notably certain light alloys, have no true endurance limit; their S-N diagrams never become quite horizontal. The endurance strength for any number of cycles can be ascertained from the S-N diagram, if available. If the endurance strength s, for a number of stress cycles n, is known, then the endurance strength s_2 for n_2 cycles can be estimated by using an assumed equation for the S-N diagram.

A solenoid valve has a repeated stress governed by the degree of impact between poppet and seat. Materials under repeated stress conditions exhibit a cycle-life corresponding to the degree of stress. Therefore, the life of a solenoid valve is governed by the impact stress and the metallurgical effects from repeated stressing of the applicable materials used in the poppet and seat. The effects of the repeated impact stress on the life of a solenoid valve are given in the following discussion.

3. RELIABILITY ESTIMATE FROM IMPACT STRESS AND THE EFFECTS OF REPEATED STRESS

The foregoing discussion on repeated stress leads to the assumption that an induced stress can be used with the applicable S-N diagram for the prediction of the life expectancy of a mechanism.

A survey of published literature shows that industry's approach to the prediction of the expected life of mechanical parts has been by theoretical analysis and that the results obtained by this method are favorable.

During the survey an attempt was found to have been made in predicting the expected life of gears by applying the compressive stress between mating gear-teeth to applicable S-N diagrams. The diagrams were for materials in compressive, nonreversed loading. The compressive stress created by the dynamic loading between mating gear-teeth was calculated from a theoretical analysis similar to the one contained in the discussion on impact stress presented in Section 2.2.2. However, the compressive stresses used in this life-prediction attempt were modified by factors that considered the lubrication between teeth, mating surface conditions, and the galling tendency of mating materials where the modification factors were determined by extensive laboratory tests. Furthermore, the S-N diagrams were used to indicate failure of the mating surfaces (pitting, spalling, or flaking off of material), and, therefore, excessive backlash. Complete failure, or breaking-off of the teeth, was assumed to follow in a very short period of time (1 to 3 hours).

The results of the above attempt showed that the compressive stress, modified by the appropriate modification factors and the applicable S-N diagram, yielded an accurate estimate on the expected life of a gear when compared to actual test results. In the ARINC Research study, the inability to close, or seal off, the pressurized gases to the outside environment is assumed to constitute failure of a solenoid valve. A valve's ability to seal off a gas depends on continuous consistency of the surface conditions of the poppet and seat sealing-areas throughout operation. The first indications of stress-cycle failure are minute surface cracks, spalling or flaking away of surface material, and permanent deformation.

Therefore, the impact stress versus the applicable S-N diagram can be assumed to give an approximate prediction on the expected life of a solenoid valve. For example, a valve with the seat and poppet made of 2024-T4 aluminum alloy, which exhibits an impact stress of 55,000 psi would, as calculated from equation (12), have a life of approximately 16,000 cycles from the applicable S-N diagram. See Figure 8.

Ordinarily, the above hypothesis would have been proved valid by a comparison of numerous different valve applications with corresponding life test data. This, however, would have involved a time requirement far in excess of the time allotted to this study. Consequently, only a small number of valves were theoretically compared with the actual life test data. In all instances, the theoretical life prediction was in close agreement with the actual life. For valves with an actual life of 20,000 cycles, the results were within 3,000 cycles; and, for valves with an actual life of 1,500,000 cycles, they were within 200,000 cycles. The assumed stress area and length, unknown surface conditions, and cold working are believed to be responsible for the error.

When the above method is used for a poppet and seat combination made of two different materials, it must be remembered that the impact stress is to be used with the S-N diagram of the material which yields the least cycles-to-failure because the impact stress, equation (12), is based on the modulus of elasticity of two bodies.

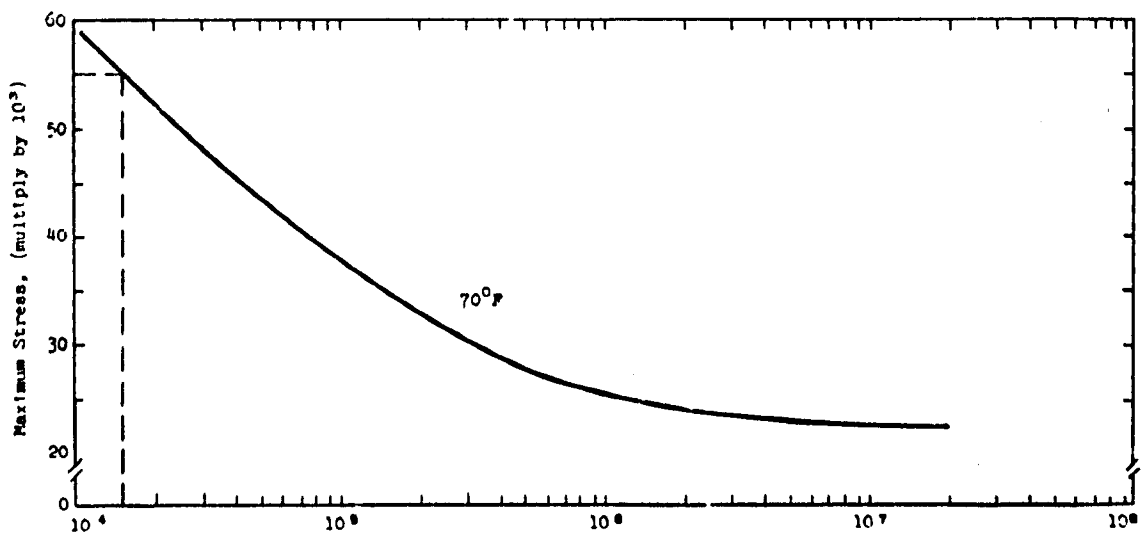

FIGURE 8

S-N CURVE OBTAINED FROM LIMITED TESTS OF A TYPICAL GROUP OF 2024-T4 SPECIMENS AXIALLY LOADED, STRESSED IN COMPRESSION

4. THE VARIABLES AFFECTING THE RELIABILITY OF THE SEAT AND POPPET

As stated previously, all solenoid valves may exhibit a typical expected life, but, after compiling and evaluating data, solenoid valves were found to exhibit lives dependent upon the particular application. The discussion on impact and repeated stress, Section 2.2.2 and 2.2.3, supports the statement that the life depends on the particular application since many variables can affect the impact stress between the poppet and seat. Therefore, an analysis of some of the important variables that can reduce the expected life during operation or cause an erroneous estimate of the expected life is necessary. Although the variables are obviously numerous, the major and most critical ones are: poppet velocity; poppet travel; material used in the poppet seat; pressure differential; temperature; and space application.

4.1 Poppet Velocity

Since equations (3) and (4) use time (t) and distance (d), the velocity (v) at the instant of impact can be found for equation (12) from the equation v = 2d/t. Thus, a decrease in time causes an increase in velocity for equation (12), thereby increasing the impact stress between the poppet and seat.

The above statement and a typical S-N diagram demonstrate that the expected life of a valve will decrease/increase with the decrease/increase of the response-time-to-close.

For example: when the equivalent (2d/t) is substituted for v in equation (12), then

$$S = \frac{1}{2}\left\{\left[\frac{F_s+F_p}{A} + \left[\left(\frac{F_s+F_p}{A}\right)^2 + \frac{16d^2W_pE_1E_2}{t^2gA(L_1E_2+L_2E_1)}\right]^{1/2}\right]\right\}$$

If all variables, except the response time (t), are held constant in the aforementioned equation, and (t) is decreased, then the impact stress is increased (see Figure 8) thereby yielding a progressively shorter life-cycle. Furthermore, when all variables are held constant in a valve application, except the response time (t), the variation in the life cycle corresponds to that in the response time, as shown in Figure 9.

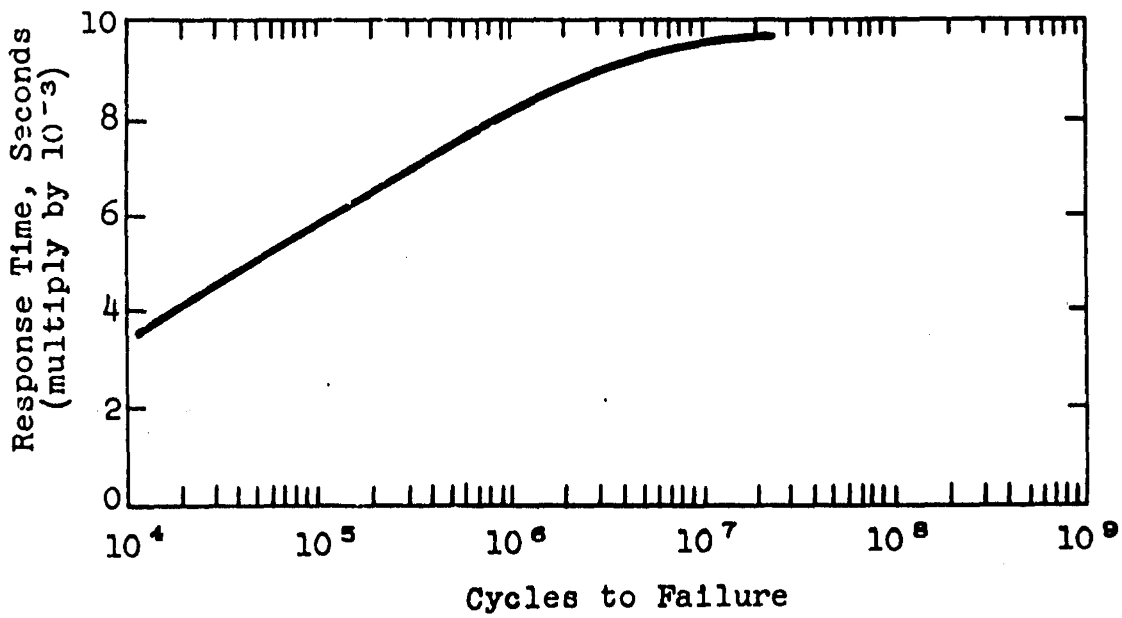

FIGURE 9

RESPONSE TIME VS. CYCLES TO FAILURE OF A TYPICAL SOLENOID VALVE DESIGN WITH ALL OTHER VARIABLES CONSTANT

4.2 Poppet Travel

The distance traveled by the poppet from the full open position to the closed position constitutes the distance (d) in equation (12). Thus, with all other factors held constant except (d), an increase in distance will decrease valve life, while a decrease in distance will increase valve life. This statement can be substantiated by the use of the procedure outlined in the discussion on velocity, Section 4.1.

An example of the effect of poppet travel on valve life is shown in Figure 10.

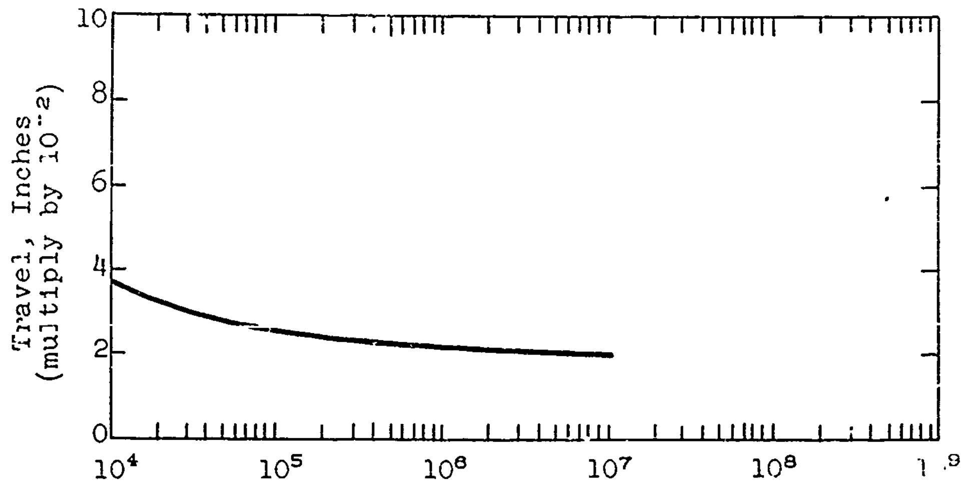

FIGURE 10

POPPET TRAVEL VS. CYCLES TO FAILURE
OF A TYPICAL SOLENOID VALVE DESIGN
WITH ALL OTHER VARIABLES CONSTANT

4.3 Temperature vs. Reliability

Fatigue tests of materials have proved that the cycle-life varies inversely with the temperature. Figure 11 shows the S-N curves for a typical material application at various temperatures. It also demonstrates that, when the impact stress between a poppet and seat is calculated at a temperature of T_1, and the temperature rises to T_2, with the impact stress held constant, the increase in temperature shortens the life. It is necessary to use the material S-N curve that has been obtained at a temperature approximating that which the control valve will experience while in orbit.

4.4 Static Load

Figure 12 shows that on the poppet and seat of a valve, a relationship exists between the static load, the leak rate, and the reliability. Although the curves shown are generalized, because no specific data were available, it is theorized that a reliability trade-off exists between the three variables presented above, and it can be made upon proper analysis.

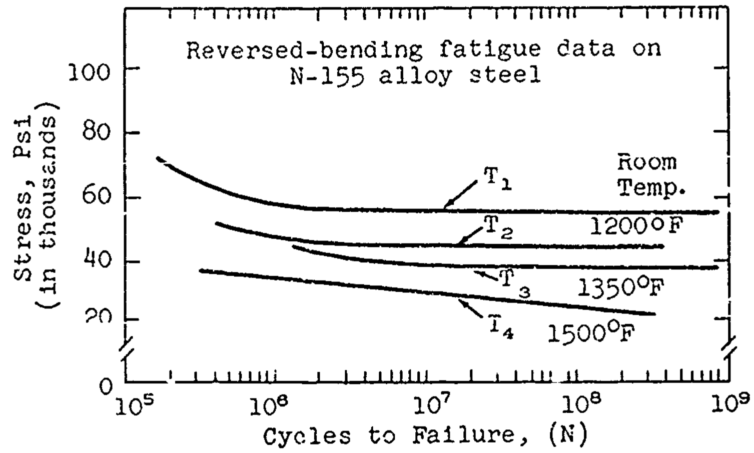

FIGURE 11

THE EFFECTS OF TEMPERATURE CHANGE
ON A MATERIAL IN REPEATED STRESS

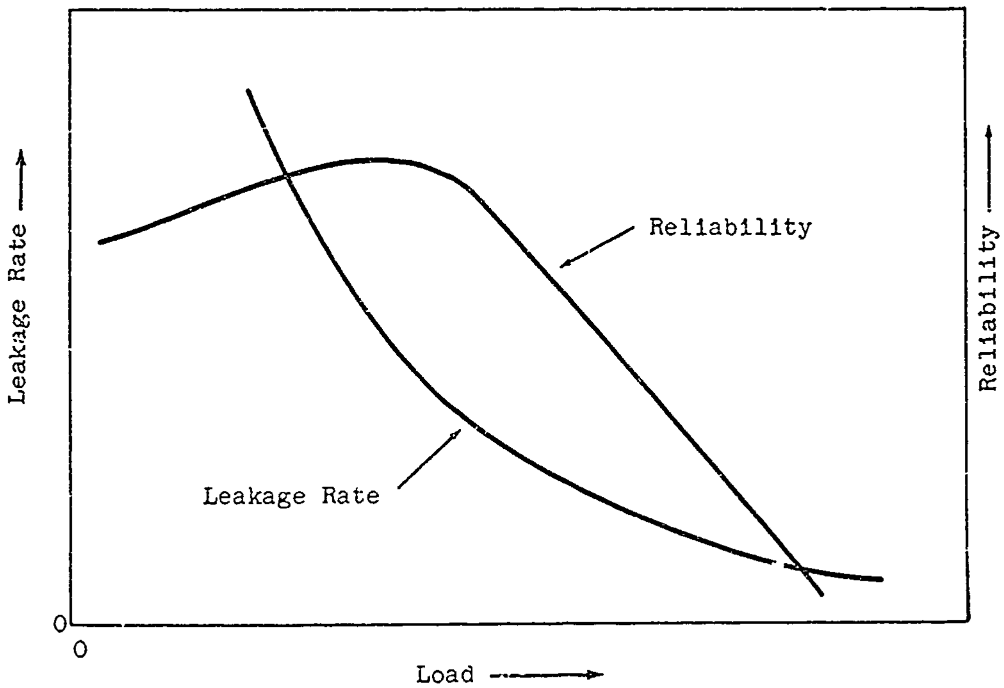

FIGURE 12

TRADE-OFFS AFFECTING RELIABILITY, LEAK RATE, AND LOAD

As might be expected, examination of the load curve reveals that, as the load between the poppet and seat is increased, the leak rate will decrease to some point beyond which any further load increase results in no measurable leak-rate decrease. It is, of course, desirable to load the poppet as much as practicable in order to produce a valve with the smallest possible leak rate, but this may lead to the life-cycle problems discussed in the following paragraphs.

The reliability curve illustrates that an initial increase in reliability will result when the load is enlarged, but that this increase is not a monotonic function. On the contrary, a point exists where a further increase in the static load between the poppet and seat will result in an ever-decreasing reliability. The shape of the reliability curve can be explained as follows: As the load is increased, the observed reliability first increases because the leak rate is graduallly being reduced; however, the reliability curve soon reaches an apex where it remains constant for a period and then, as more load is applied, it decreases or falls off rapidly. This occurs because the increase in load produces local stresses that fatigue the material used for the poppet and seat. Thus, to withstand cycling, the valve should have the smallest possible leak-rate which, in part, is a function of the static load. On the other hand if the static load is extremely large, then the number of cycles that the valve can reliably perform will be relatively small. Therefore, it is necessary to know, within reasonable limits, either the number of cycles that the valve will be required to operate or the leak rate which will be acceptable so that a trade-off can be made that will effectively produce the optimum reliability.

4.5 Space Application

4.5.1 Precycling

ARINC Research has defined one of the failures of the solenoid valve as the inability of the poppet and seat to effectively seal the contained gases in a pressurized system from the outside environment. The analysis of the compiled data acquired for this study revealed that a solenoid valve exhibits a continuous leakage rate at an outlet pressure approaching the vacuum conditions of outer space. An average of the leakage rates versus the number of cycles at various inlet pressures and approximate outer space vacuum-condition outlet-pressures is shown in Figures 13 and 14 for solenoid valves having metal to metal and nonmetal to metal sealing areas.

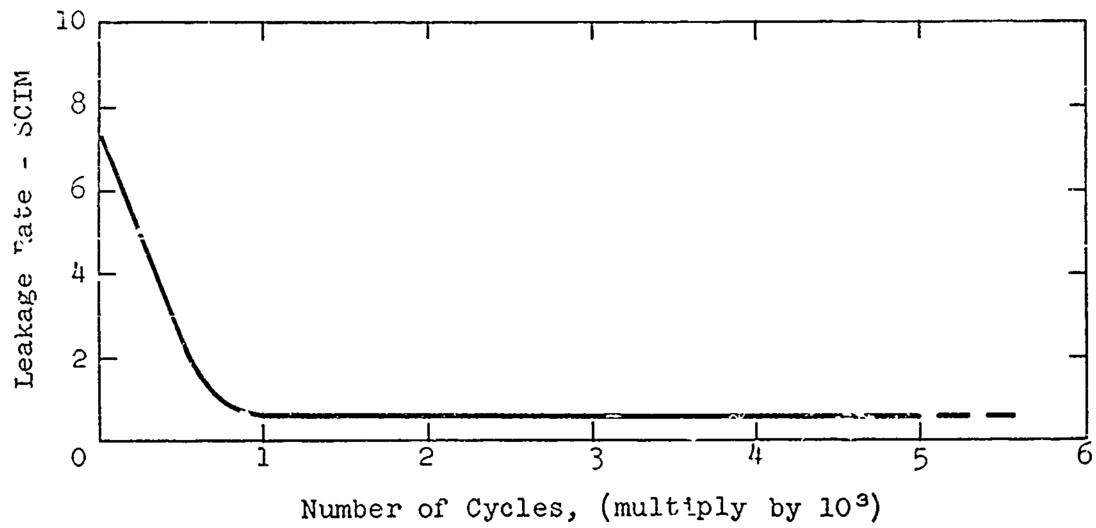

FIGURE 13

AVERAGE SOLENOID VALVE LEAKAGE RATE VS. NUMBER OF CYCLES WITH METAL TO METAL SEALING, AVERAGE OF VARIOUS INLET PRESSURES, AND APPROXIMATE 0 PSIA OUTLET PRESSURE

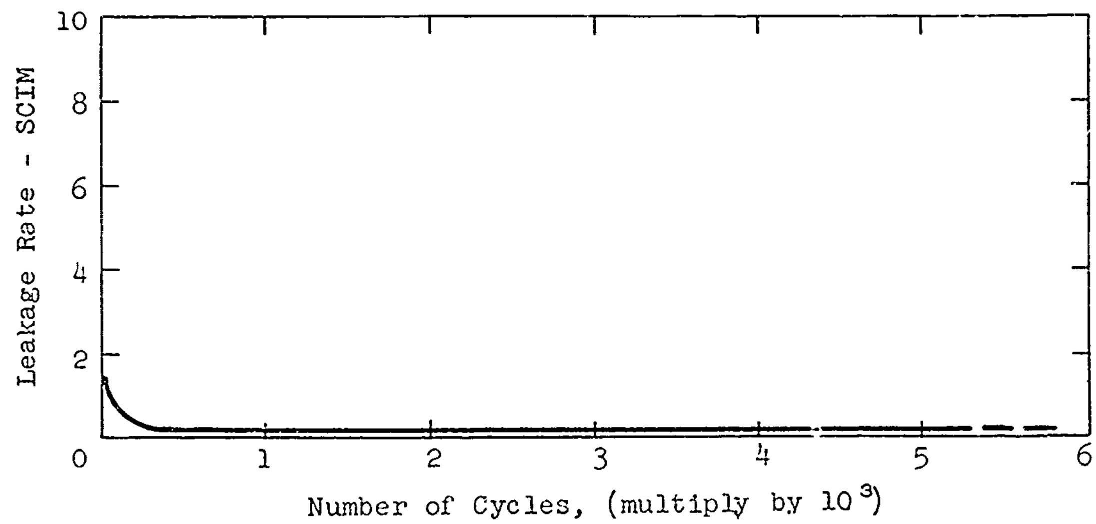

FIGURE 14

AVERAGE SOLENOID VALVE LEAKAGE RATE VS. NUMBER OF CYCLES, WITH NONMETAL SEALING, AVERAGE OF VARIOUS INLET PRESSURES, AND APPROXIMATE 0 PSIA OUTLET PRESSURE

The main purpose of Figures 13 and 14 is to show the effects of "wear-in" on the leak rate of a poppet and seat. As might be expected, the effects of wear-in are more pronounced for the metal-to-metal than for the nonmetal-to-metal poppet and seat. This is, more or less, an intangible aspect of poppet and seat reliability but an important consideration, nevertheless, for, by precycling the valve, the leak rate can be stabilized within close limits, leaving only the effects of wear to increase it.

4.5.2 Materials

The effect of the space environment on the materials of solenoid valve-parts is difficult to ascertain. However, one known phenomenon that may directly affect the life of a solenoid valve is the evaporation of materials due to a vacuum. If the materials making up the poppet and seat should evaporate at a high rate, the results would be obvious. Figures 15 and 16 show the effects of a vacuum on some metals and nonmetals.

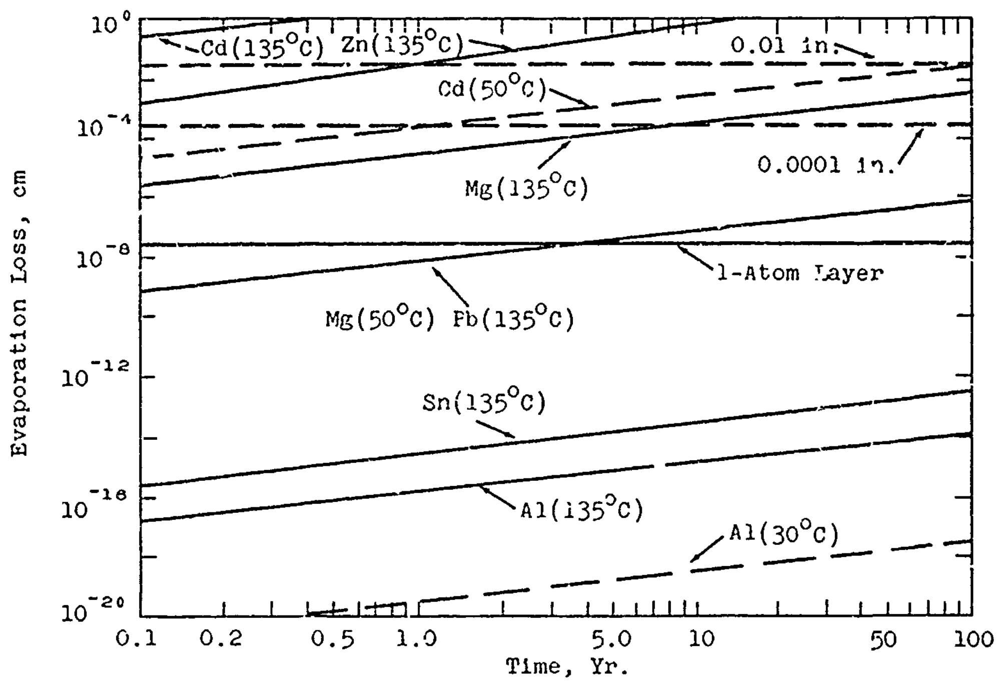

FIGURE 15

THEORETICAL EVAPORATION OF SOME METALS IN A VACUUM AT APPROXIMATELY 10^{-9} TO 10^{-10} MILLIMETERS OF MERCURY

Polymer	% Loss After 24 Hrs. at 300°F	Polymer	% Loss After 24 Hrs. at 300°F
Kel-F	1.6- 2.2	Linseed Oil	10.5-28.6
Silicone	2.3- 6.6	Acrylic	17.5-42.8
Cellulose Acetate Butyrate	2.3-11.2	Polyester	19.7
Styrene Butadiene	2.7- 4.7	Neoprene	26.3
Phenolic	4.5	Nitrocellulose	47.2
Epoxy	7.3-28.4	Vinyl Copolymer	60.3
Polyvinyl-Butyral	8.7-41.7	Polysulfide	53.1-63.5
Alkyd	9.8-17.4	Polyurethane	66.5
Silicone-Alkyd	10.0-13.7	Nitrocellulose	82.3

Source: 1st Symposium "Surface Effects on Spacecraft Materials" Ed. by Clauss

FIGURE 16

WEIGHT LOSSES FOR SOME NONMETALS IN A VACUUM AT APPROXIMATELY 10^{-9} TO 10^{-10} MILLIMETERS OF MERCURY

The materials listed in Figures 15 and 16 are seldom, if ever, used in the fabrication of valves. However, because they are vulnerable to the evaporation phenomenon a reasonable assumption is that the materials used in valves would also be affected. If this hypothesis is correct, then a safety margin should be added to the calculated gas supply. ARINC Research has been unable to ascertain the magnitude of the percentage which should be used as a safety factor, but an increase in the gas supply of somewhere between 6-8% is believed to be satisfactory provided the materials utilized in the valve are stainless steel and/or aluminum.

5. COMBINED RELIABILITY

5.1 Statistical Aspect

The previous sections have been mainly concerned with the development of a theoretical technique for predicting the reliability of the poppet and seat combination. Although these play a large role in valve reliability, other part reliabilities must also be considered. These parts, as discussed in Section 2.1, are: electrical solenoid, spring, poppet and seat.

In order to predict the reliability of a solenoid valve, the reliabilities of the parts described above must be combined; but, before this can be done, the failure-density function, the mean lives, and/or the failure rates of these parts must first be determined so that the failure-density function can be properly combined. The end result of this reliability approach produces a probability-of-survival curve for a specified period of time. ARINC Research Corporation favors the probability-of-survival method for displaying mechanical reliability results because this type of presentation affords flexibility.

5.2 Part Reliabilities

5.2.1 Solenoid and Spring

The reliability aspects of the solenoid and spring have been combined in this section because each is expressed as a failure rate and each is assumed to be best described by the exponential failure-density function. The purpose of assigning a failure rate to the spring is simplification of the realiability prediction. Ordinarily, the failure-density function of a spring is normally distributed because it results from a fatigue-failure phenomenon. This simplified exponential assumption is justifiable because most springs are designed to operate within the endurance limits of the spring materials and, therefore, fatigue failures are not likely to be observed until about 10^7 - 10^9 cycles of operation. These cycles-to-failure are well beyond the expected life of most valves.

A failure rate is assigned to cover the small probability that the spring has been improperly manufactured or inspected and, hence, will exhibit an early catastrophic failure. See Table 1.

TABLE 1

ESTIMATED FAILURE RATES AND MEAN LIVES OF SOLENOID VALVE PARTS

Part Type	Number	Estimated Failure Rate (Failures/Hr.)	Estimated Mean Life (Cycles)	Estimated Mean Life (Hours)
Solenoid	1	36×10^{-6}		
Spring	1	5×10^{-6}		
Poppet & seat	1		16,000*	16[illegible]**

* Computed as described in Section 3.

** Hypothetical estimate of 10 cycles per hour. (This should be the same estimate used to calculate attitude control gas supply.)

The electrical solenoid failure-rate, shown in Table 1, is an ARINC Research published figure which was obtained from data collected in the field. These data are best described by the exponential failure-density function and, since the solenoid (discounting the armature) has no moving parts, this seems reasonable.

5.2.2 Poppet and Seat

The method developed in Section 3 for predicting the life of a poppet and seat yields an answer in terms of a mean life. It does not take into account the variability of occurrence, which is a function of the type of distribution associated with the particular wear-out phenomenon. It is generally agreed that the wear-out phenomenon of simple devices can be best described by the normal distribution and this is hypothesized to be true also of the poppet and seat. The standard deviation, ($\sigma = 7{,}200$) is a best estimate resulting from an

analysis of fatigue-failure data of materials recently investigated by ARINC Research in an earlier mechanical-wear study.

As mentioned in Section 5.1, the individual part reliability must be combined to reflect a true picture of the entire device. Table 1 summarizes the part reliability data presented in Sections 3 and 5.2.2 for the solenoid valve. It shows that two exponentially distributed functions must be combined with one normally distributed function. The analysis of the examples provided, which are illustrated in Figures 17, 18, and 19, was made by a simple, straightforward method that follows the conventional pattern of analysis used by most electronic reliability-prediction groups. The probabilities of survival for the parts are computed as shown in Figures 17 and 18. The overall reliability function, R(t), for the entire solenoid valve is obtained as a function of the equation presented in Figure 19.

$$R = e^{-\lambda_1 t} \cdot e^{-\lambda_2 t} = e^{-t\Sigma\lambda_i}$$

$$\Sigma\lambda_i = \begin{array}{l} 5.0 \times 10^{-6} \\ +\,36.0 \times 10^{-6} \end{array} \text{ or } 41.0 \times 10^{-6}$$

$$R = e^{-.000041t}$$

t	.000041t	$e^{-.000041t}$
0	0.000	1.000
4000	0.164	0.935
8000	0.328	0.720
12000	0.492	0.612
16000	0.656	0.519
24000	0.984	0.386
32000	1.312	0.271

FIGURE 17

RELIABILITY CONSIDERING ONLY PART FAILURES BY AN EXPONENTIAL DISTRIBUTION (Springs, Solenoids)

$$R = P_s = 1 - \int_{-\infty}^{Z} f(t)\,dt$$

where

$$\int_{-\infty}^{Z} f(t)\,dt = P_f$$

Mean (μ) = 16000

σ = C.V. x μ

σ = .45 x 16000

σ = 7200

	Z	P_f	
t	$\frac{t-\mu}{\sigma} = \frac{t - 16000}{7200}$		$P_s = 1.0 - P_f$
0	$\frac{-16000}{7200} = -2.222$	.0131	0.987
4000	$\frac{-12000}{7200} = -1.666$	.0479	0.952
8000	$\frac{-8000}{7200} = -1.111$	.1333	0.867
12000	$\frac{-4000}{7200} = -0.555$	.1894	0.811
16000	$\frac{0}{7200} = 0.000$	0.500	0.500
24000	$\frac{8000}{7200} = 1.111$	0.8667	0.133
32000	$\frac{16000}{7200} = 2.222$	0.9869	0.013

FIGURE 18

RELIABILITY CONSIDERING ONLY PART FAILURES DESCRIBED BY A NORMAL DISTRIBUTION
(Poppet and Seat)

t	Exponential	Normal	R
0	1.000	0.987	0.987
4000	0.935	0.952	0.890
8000	0.720	0.867	0.624
12000	0.612	0.811	0.496
16000	0.519	0.500	0.260
24000	0.386	0.133	0.051
32000	0.271	0.013	0.004

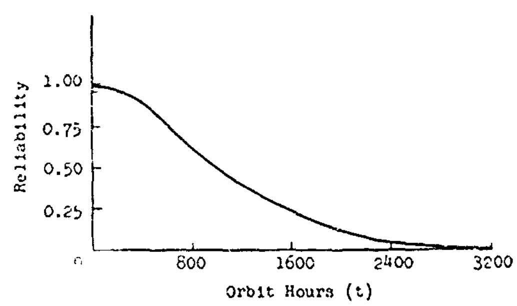

FIGURE 19

COMBINING OF PART RELIABILITIES TO GIVE SOLENOID VALVE RELIABILITY

6. CONCLUSIONS AND RECOMMENDATIONS

6.1 Conclusions

A technique is presented for determining the reliability of solenoid valves ~~as a result of this initial study~~. ~~ARINC Research Corporation~~ It is believed this effort was ~~to be~~ a step ~~in the right direction~~ toward the prediction of the reliability of mechanical and electromechanical devices. However, a detailed empirical analysis should be performed in the near future to either refute or substantiate the theoretical modes of failure analysis as well as the theoretical prediction technique ~~presented~~ ~~in this report here~~.

It is p... The present effort did not include this highly desirable aspect because of time limitations. An empirical analysis should include considerations such as the effects of vacuum conditions, temperature, wear compatibility between unlike materials, and stress concentration between seat and poppet.

6.2 Recommendations

As a result of this study, ARINC Research Corporation recommends the following procedure for achieving maximum valve reliability:

1. Precycle the valve
   - (a) 100 cycles for metal-to-metal sealing
   - (b) 20 cycles for metal-to-nonmetal sealing
2. Pretest the valve to determine the actual leakage rate, simulating the conditions of outer space environment as nearly as possible.
3. A cold gas reaction attitude control system should be designed so that the solenoid valves have the slowest possible response time because a fast response time causes high stresses and thus a short life.

4. The reaction gas should have a minimum moisture content to prevent to the extent possible, the formation of ice between the poppet and seat.

5. The use of nitrogen as the reaction gas is recommenued because it exhibits lubricating characteristics.

6. The complete attitude control system should be free of all foreign matter and particles. The use of filters is also recommended.